Effect of Damping Element Damage under Erosion on Vibration Behavior of an Industrial Gas Turbine Group-Blades

Seyed Ahmad Mortazavi[1], Abbas Rahi*[2], Seyed Mohammad Jafari[3]

[1] Ph.D. Candidate, Faculty of Mechanical and Energy Engineering, Shahid Beheshti University, Tehran, Iran
E-mail: a_mortazavi@sbu.ac.ir

[2*] Associate Professor, Faculty of Mechanical and Energy Engineering, Shahid Beheshti University, Tehran, Iran
E-mail: a_rahi@sbu.ac.ir

[3] Assistant Professor, Faculty of Mechanical and Energy Engineering, Shahid Beheshti University, Tehran, Iran
E-mail: m_jafari@sbu.ac.ir,

Abstract

The last stage blade rows of modern low-pressure gas turbines are subjected to high static and dynamic loads. The centrifugal forces primarily cause the static loads due to the gas turbine's rotational speed. Dynamic loads can be caused by stationary gas forces, for example. A primary goal in designing modern and robust blade rows is to prevent high cycle fatigue caused by dynamic loads due to synchronous or non-synchronous excitation mechanisms. Damping elements are one of the most common structures to alleviate excessive vibration amplitudes in turbo machinery applications. This paper deals with fracture investigations of the gas turbine blade of a 15 MW Gas injection station in national Iranian south oil company in the southwest of Iran. Macroscopic and scanning electron microscopy images of the fracture section of the tube show two phenomena erosion and fatigue. Therefore, to more accurately identify the cause of the failure, stress and vibration analysis of the blade is performed individually and coupled with other blades by the connecting tube using ANSYS software. To validate finite element results, modal test of single blade and group of blades is done. According to the observation of fatigue at the section of the tube failure and the possibility of error in the design, the sensitivity measurement of the diameter and installation position of the tube is done.

Keywords: Damping element, Vibration, Modal test, Gas Turbine Blade, Erosion.

1. Introduction

Particularly broad operating ranges and the appearance of a multitude of excitation mechanisms as present in industrial turbomachinery like turbines, compressors, pumps, fans, etc. require a resonance-proof blade design. Above all variable speed applications are associated with continuous resonance situations, which must not compromise the mechanical integrity Schubert and Tusche [1].

Low-pressure turbine blades (LPTB) are exposed to very large centrifugal and aerodynamic forces during operation and their stiffness is quite low, which makes them to operate in a very severe condition. Relatively thin airfoils with complex 3D shapes need to be utilized in order to maximize efficiency and to minimize kinetic energy losses. In addition, the high variability of operating conditions, which is a typical situation for mechanical drive applications, requires different types of damper design approaches to be able to increase effectiveness. There are several studies for the nonlinear dynamic assessment of LPTBs with friction interfaces in the literature [2-5].

One of the effective ways to prevent blade resonance is to vibration control or keep the vibration amplitude low. For constant velocity applications, this is done by tuning the blade frequencies separate from the engine order harmonics. In contrast, in several industrial applications, turbines have to run with variable speed – e.g. for mechanical drives – and blade resonance vibration will occur when a blade natural frequency coincides with one of the engine order harmonics. In the case of resonant vibration, a practical and straightforward way to reduce vibration amplitudes is to introduce damping elements like damping tubes or lacing wires, which use energy dissipation from dry friction [3]. The most common examples are the blade-disk interfaces [6, 7], the shrouds at the blade tip [8, 9], ring dampers [10, 11] and friction dampers located under the blade platforms [12-14]. One of the important solutions to reduce dynamic response amplitudes is to avoid resonance regions as much as possible with blade detuning. However, changing the natural frequency values of the turbomachinery parts is not too feasible due to the high modal density and wide spectrum of the external excitation force. Another very well-known and efficient technique is to utilize dry friction damping for passive control of the vibration. There are several ways used in the turbine blades by deliberately implementing friction to the system in order to dissipate the energy [5]. Various research works have been done on the effects of erosion and corrosion particles in the hot gases passing through the turbine blades. In near-sea installations, chloride salts are very common compounds found in the air flowing to the turbine. Such corrosive and abrasive factors significantly damage the damping elements between the blades. The damage is such that pitting is formed on the surface of the damping element, which becomes the place of stress concentration, crack growth, and failure under the effect of heavy loads caused by centrifugal and aerodynamic forces [15-19]. You [20] established a method for predicting the fatigue life of the blades considering external loads, employing the theory of continuous damage mechanics. Qi et al. [21] calculated the steady and unsteady flow fields to study the blade vibration characteristics. They derived the variation law of aerodynamic damping with the blade phase angle and blade local working coefficient under critical conditions. Fuhrer et al. [22] used the energy method to predict the aerodynamic damping of the last stage blade of the steam turbine through numerical simulations. They compared the potential flutter hazards of the aerodynamic damping and different blade root fixation methods under different traveling wave modes. Rzadkowski et al. [23] simulated the aeroelastic problem of the last stage rotor blade of the steam turbine. They obtained the low-frequency component of the axial force. The results showed that the unsteady pressure fluctuation would cause the rotor blade to vibrate in different bending torsion modes. Wang et al. [24] derived the closed-form formulas for damping strength and resonance frequency and proposed a vibration theory method that can obtain the resonance frequency of nonlinear oscillations. Graciano et al. [25] analyzed the damping, stiffness, and natural frequency of steam turbine blades using Ansys software. Based on the obtained results, the Steinberg 3-band technique and Palmgren-Miner cumulative damage rule were used to calculate fatigue damage and useful life. Wang et al.[26] investigated the vibrations of the turbine blade with cracks using a nonlinear dynamic vibration analytical model and investigated its effects on the fatigue characteristic. Hu et al. [27] investigated the vibration characteristics and blade life estimation of the last row of the steam turbine under the effect of damage caused by water droplets. Previous studies have mainly investigated mistuning due to blade damage. However, in practical applications, it should be noted that the damping element often damages in blade disk system. Few works [28-30] have been focused on bladed packets due to shroud damage. Distributed parameter model is applied to investigate the effect of lacing wire damage-induced mistuning on modal characteristics [30] In order to research the effect of damage on the natural characteristics of bladed packet, an index is introduced. In order to establish the relationship between the damage and shift in the modal of mistuned bladed packet due to shroud and blade damages, a lumped parameter model is developed [29]. The shroud used to reduce vibration localization has been researched [28]. Other ways of reducing the vibration of the structure are conducted [31] gas turbines are developing towards high power and high reliability. Despite extensive research on the turbine blade, the vibrations of the coupled blade group and damage sensitivity on the damping element on an industrial turbine model have not been investigated. Some recent research has been conducted to investigate the vibrations of simple blades in laboratory setups [32].

In this paper, the vibration behavior of a group of blades of an industrial gas turbine which are coupled with a damping tube was investigated. Damping element damage, although not as catastrophic as blade failure, is more frequent in occurrences and often acts as a precursor to subsequent blade damage. Detection of damping element damage is therefore equally important. Stress analysis was performed using ANSYS software in order to identify the area of stress concentration on the damping tube. The modal test of a single blade and a group of blades is performed. Campbell's diagrams of group blades are extracted in healthy and damaged damping tubes. In the following, damping ratios and dynamic stress analysis are calculated. The FEM simulation results showed that high stress occurred in the areas where the erosion pits in macrostructure and crack initiation in microstructure under both dynamic and static forces. In the following, the damping tube effect on natural frequencies of group blades is studied to control the vibrations and stay away from the excitation frequencies to prevent the occurrence of resonance. The damping tube damage leads to a decrease in the natural frequency of the first bending mode and reduces the fatigue life of the blades. The dynamic stress calculations show that the first bending mode is the most critical mode.

2. Problem definition

The free power turbine serves to convert the gas energy to mechanical rotary energy or brake horsepower. Closely coupled to the gas generator by a transition duct, the power turbine extends the gas generator combustion products through one or two turbine stages and the residual gas energy is exhausted to the atmosphere [33]. The rotative power produced in the power turbine is then available for mechanical couplings to the driven equipment. Fig. 1 shows a cross-section view of the 15MW gas turbine package that was utilized to drive a centrifugal pipeline booster to compress natural gas. Exhaust gas from the hot section of the gas generator is used as a power drive for the turbine, with a temperature of about 637 ˚C. Fig. 2 shows the general assembly of the relevant power turbine. The first stage of the power turbine consists of 42 blade sets. This stage consists of 6 groups of 6 blades and 2 groups of 5 blades with a damping tube.

Fig. 3 (the only remaining data of the desired turbine) shows the monitoring parameters by the gas station control room, which include, the gas generator and power turbine rotational speed in startup mode until reaching the nominal speed. The power turbine nominal speed is 4960 rpm, which will increase up to 5200 rpm. The over speed power turbine speed is 5500 rpm. The first stage of the power turbine blade made of nickel-based super alloy Waspaloy and the damping tube made of Inconel 718, having an operating temperature of about 550˚C, pressure of 0.2 bar and rotates 4900 – 5200 rpm was investigated.

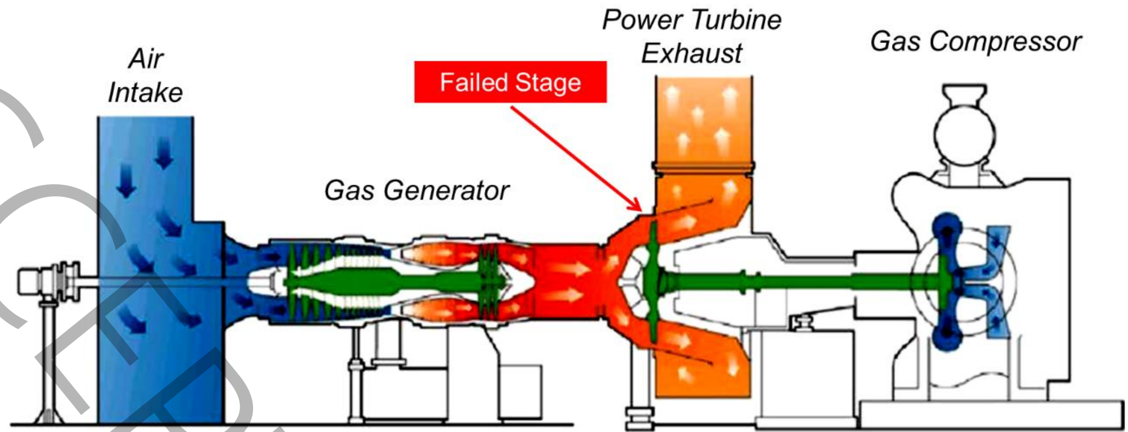

Fig. 1. Gas turbine and power turbine schematic.[34]

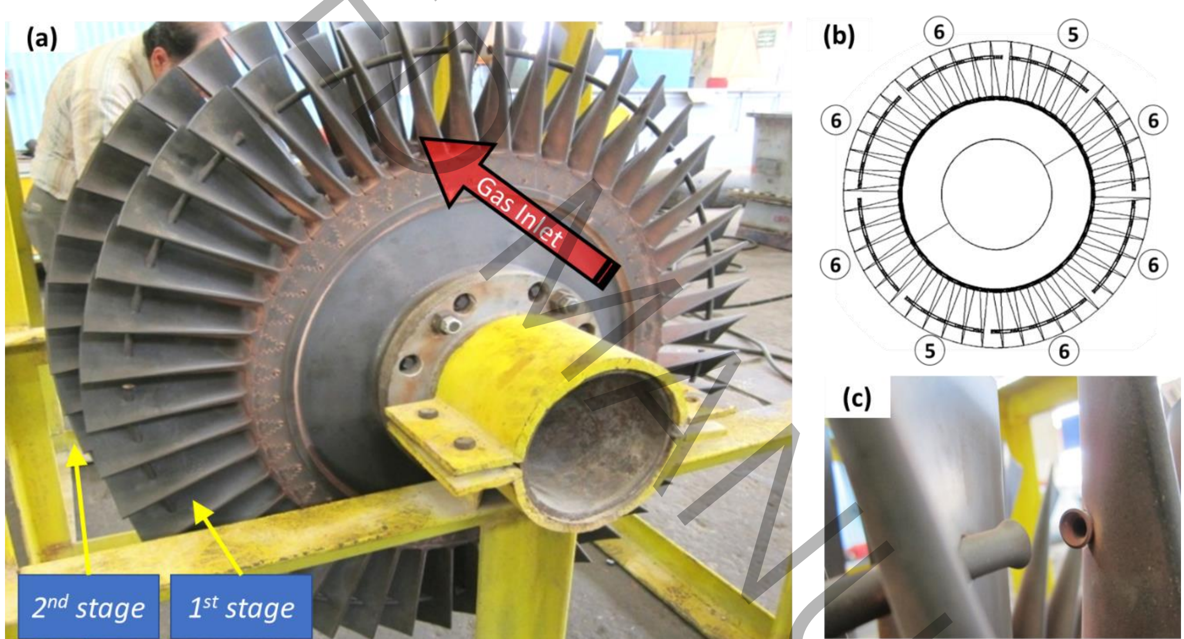

Fig. 2. The desired gas turbine (a) Rotor assembly, (b) Arrangement of a group of blades (c) Connection between damping tube and blade

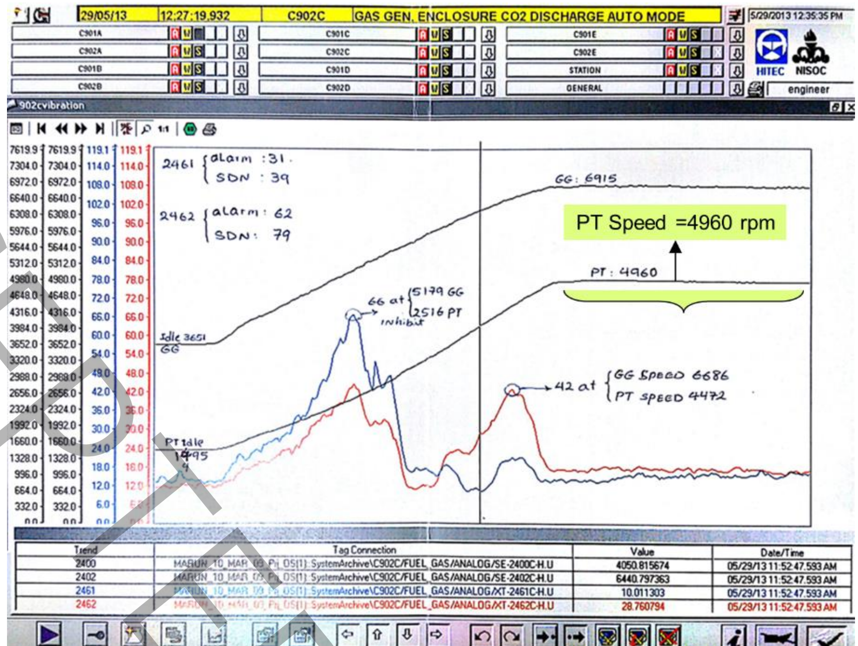

Fig. 3. Parameters recorded by control room during unit operation

After visual inspection during the unit overhaul of the power turbine rotor, a failure was observed in the area of the damping tube of the group of six blades. Therefore, a group of six blades was studied. The blade under evaluation is 250 mm in length. The macroscopic details of the failed tube were studied through visual inspection. The fractured damping tube is shown in Fig. 4. The failure of the damping tube has been observed in the group of six blades. This failure occurred in the left and right areas of the fifth blade.

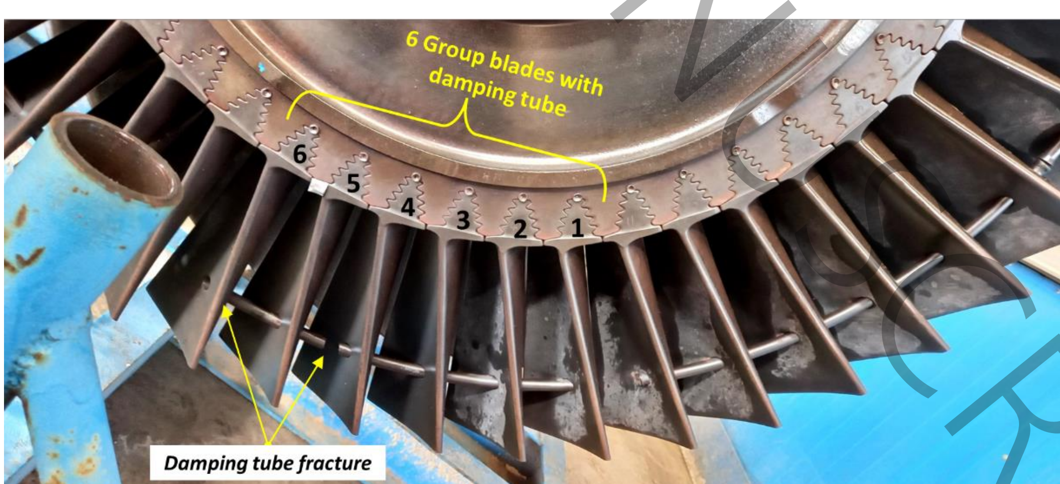

Fig. 4. The damping tube failure

The light colors in Fig. 5 indicate the eroded suction side of the blade surfaces. The origin of erosion is the separation and turbulation of the flow due to the collision with the damping tube.

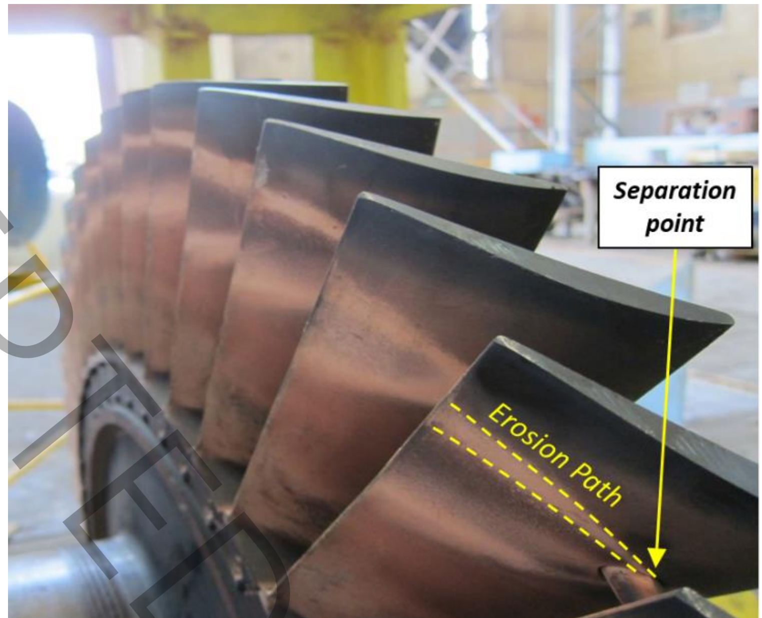

Fig. 5. Suction side eroded of first stage blades

3. Finite Element Analysis

3.1 stress analysis under static loads

The computational mechanics procedure used in this work consisted first in digitize a single blade for obtaining a file with the geometric dimensions and, secondly, to build a solid CAD model of the blade. The 3D assembly of group-blades, disk and damping tube was converted to a FEM. The 3D-modeling and assembly process shown in Fig. 6.

10-Node quadratic tetrahedron (solid 187) elements were used to mesh the blade. Analysis of a thermal-stress coupled field resulting from the collision of combustion hot gas with the blade provided accurate solutions to 3D stress coupled-field problems. This model contained 180630 nodes and 98651 elements. In defining the boundary conditions for static analysis, it was assumed that the blade was fixed on the disc and due to the thermal expansion and close compliance of the damping tube with the blade hole, it is considered a bonded connection. According to the data from the gas turbine, as received by the control room at failure time, the exhaust temperature was 637 °C, the rotor speed was 5200 rpm and the gas pressure was 0.2 MPa. As shown in to Fig. 7, the function considered for temperature distribution was a quadratic equation for blade length, and its maximum value was 637 °C.

Fig. 6. A general view of digitizing process, assembled and meshed model

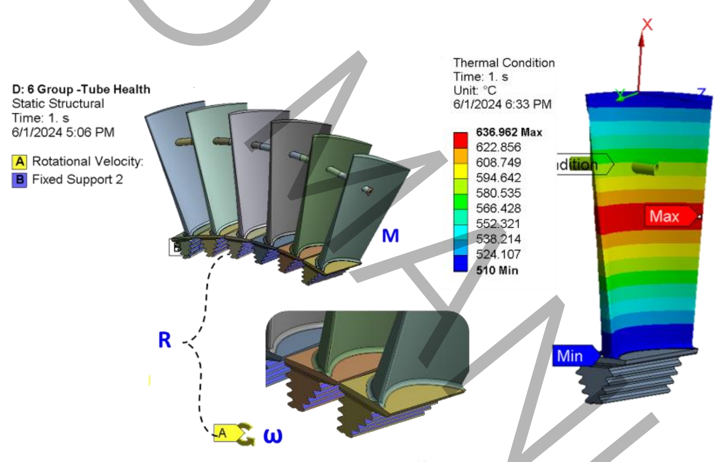

Fig. 7. Boundary conditions and loading on group blades

Afterwards, essential mechanical properties based on Table 1 are defined to software. According to the visual observation of the fracture surface and the effects of erosion on the surface of the damping tube, in general, the phenomenon of erosion can be considered as the cause of tube failure. Therefore, the static and vibration simulation of the blade group is investigated.

Based on the signs of blade erosion in Fig. 5, the pitting was first created under the erosion and wear effect due to the high velocity of the gas collision. Based on Fracture surfaces shown in Fig. 8, the micro pits on the surface of the tube in location "A" lead to the microcrack initiations around it. These micro-cracks grow up to the area of the yellow dashed line and lead to the tube failure and cutting shown in Fig. 8c.

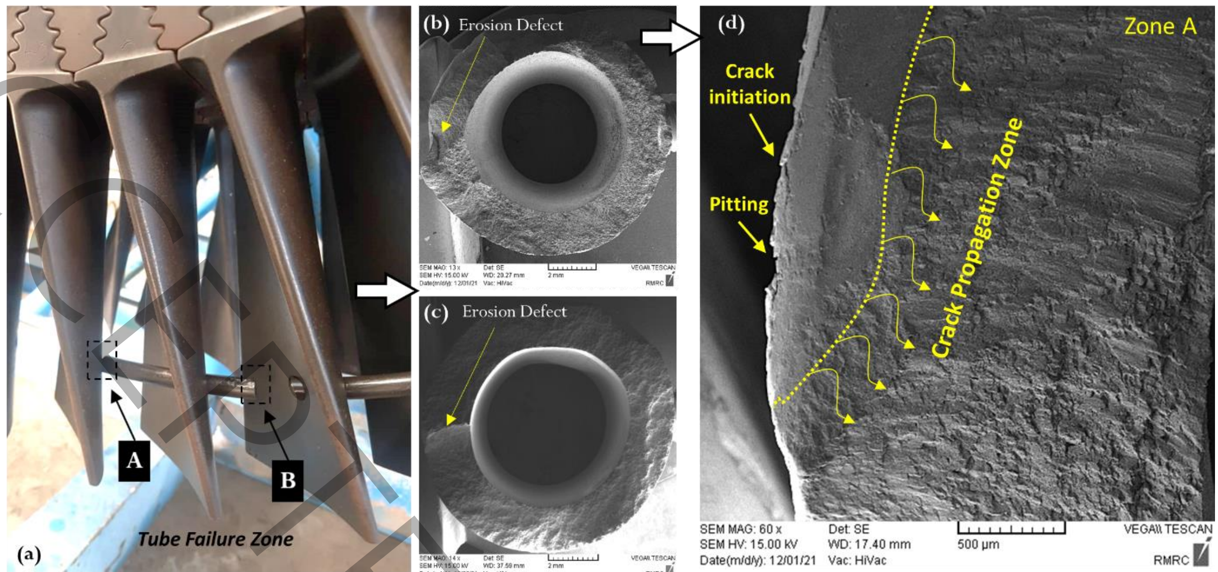

Fig. 8. Damping tube (a) failure zone, (b) Zone A (c) Zone B, (d) Magnified view of fractured surface of Zone A

Static analysis is performed in order to identify stress concentration areas and check these areas with fracture sections. The majority of the tensile stress is due to centrifugal load. The key mechanical properties were defined in the ANSYS according to Table 1.

Table 1. Mechanical properties of Blade and Tube at room and operating temperature

Materials	**Waspaloy [Blade]**		**IN718 [Tube]**	
Temperature (˚C)	21	650	21	650
Young's modulus (GPa)	211	171	208	162
Density (g/cm^3)	8.19	8.19	8.22	8.22
Yield strength (MPa)	1076	980	1172	1027
UTS [MPa]	1441	1358	1407	1158
% Elongation	27	22	21	19

The first consequence of centrifugal force is to influence the geometric form of blade. Those blades which are heavily twisted or which have non-symmetric cross sections are liable to change in their geometric form under running conditions. In the Fig. 9a, the total deformation of the group blades shows that the tube has undergone a distortion. This distortion creates areas of high stress on the surface of the tube.

Fig. 9b shows equivalent von mises stress distribution for health group blade while group-blades are connected with the damping tube and damage group blade under pitting and cutting in damping tube. Clearly, the highest stress is less than the yield strength IN718 (Table 1). Therefore, the system can sustain the forces under static conditions with no major crack. Fig. 9c shows that if the tube is cut in location A, the place of maximum stress will move to location B. Considering the closeness of von Mises equivalent stress values of health and damage tube in Fig. 9b, it is concluded that the main purpose of the damping tube was not the task of control and reduction of static stress, and its task is something else. The most important candidate parameter for the philosophy of using the damping tube between the blades is to control the vibrations of the blades, which will be discussed further.

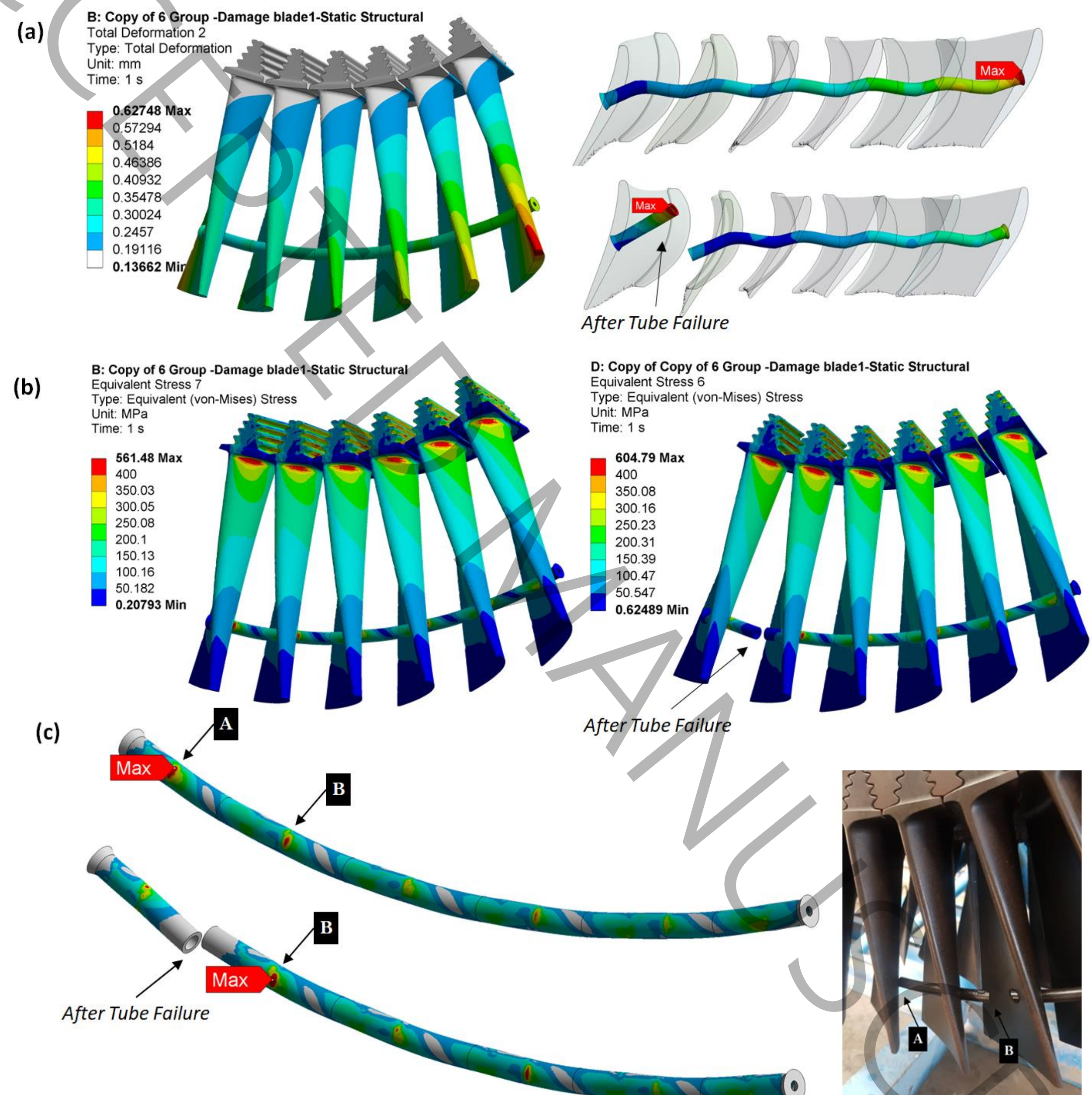

Fig. 9. Static analysis of health and damage tube (a) Total deformation of, (b) Equivalent Von-Mises stress of health and damage tube (c) stress concentration.

The pitting created by the erosion caused by the separation of the fluid flow on the surface of the tube create a high stress area close to the yield stress. This area is prone to fatigue crack growth and failure. Fig. 10 shows the Von Mises stress contour in the pitting formation area.

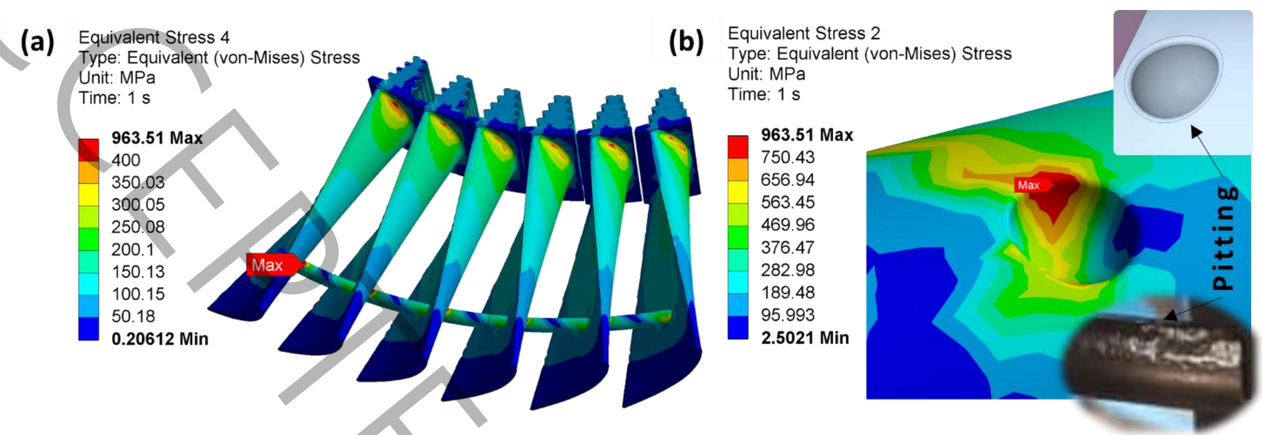

Fig. 10. Equivalent Von-Mises stress of (a) group blades, (b) pitting on damping tube

3.2 Modal and Harmonic Analysis

The resonance phenomenon increases dynamic stresses, which is the main cause of high cycle fatigue. Several factors such as manufacturing errors, erosion and corrosion can cause changes the modal characteristics of blades.

The results of a single blade frequency response are used for two purposes. First, the 3D model of the created blade is evaluated based on the complexities of the airfoil geometry and material definition. So that the difference between the FEM model and the actual blade should be minor so that the model has sufficient reliability to continue the FEM analysis. The second step, damping coefficients are extracted to calculate harmonic analysis and dynamic stresses.

One PCB-353B15 miniature piezoelectric uniaxial acceleration sensor and a 2KN impact hammer are used to perform modal testing. The comparison between the test (Fig. 11) and the FEM model (Fig. 12) shows that the frequency difference of the first mode is about 2% and the second mode is about 1%, which shows that the modeling, material definition and meshing of blade have been done with good accuracy.

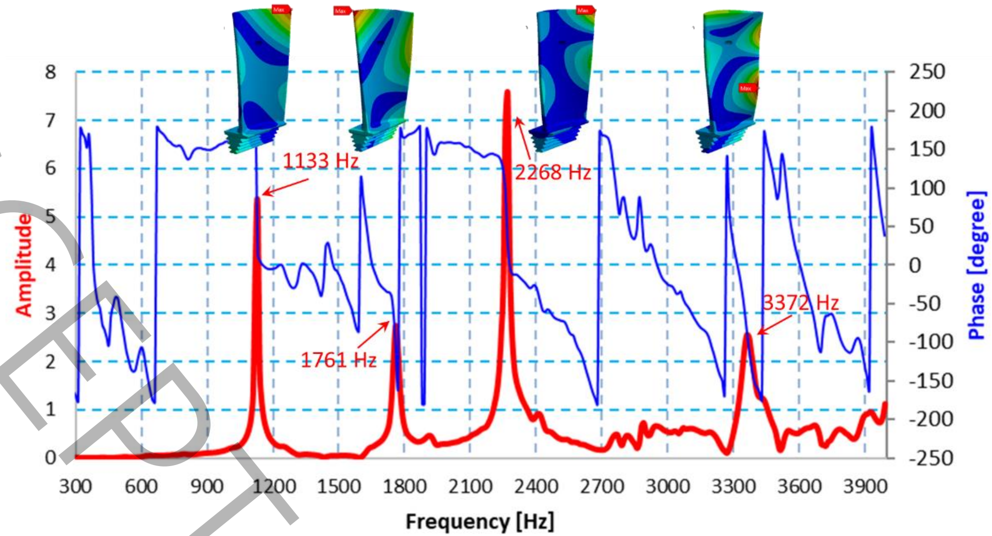

Fig. 11. FRF of free-free single blade

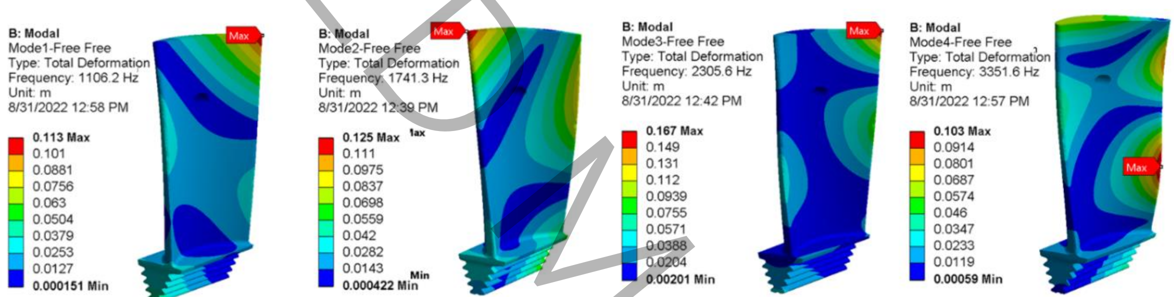

Fig. 12.FEM modal result of free-free single blade

Due to the complete assembly of the rotor and the existence of non-linear factors of surface contact, the modal test was only able to measure the natural frequency of the first bending mode of the group-blades (Fig. 13). The results of modal test and finite element simulation for the first bending mode are shown in the Fig. 14.

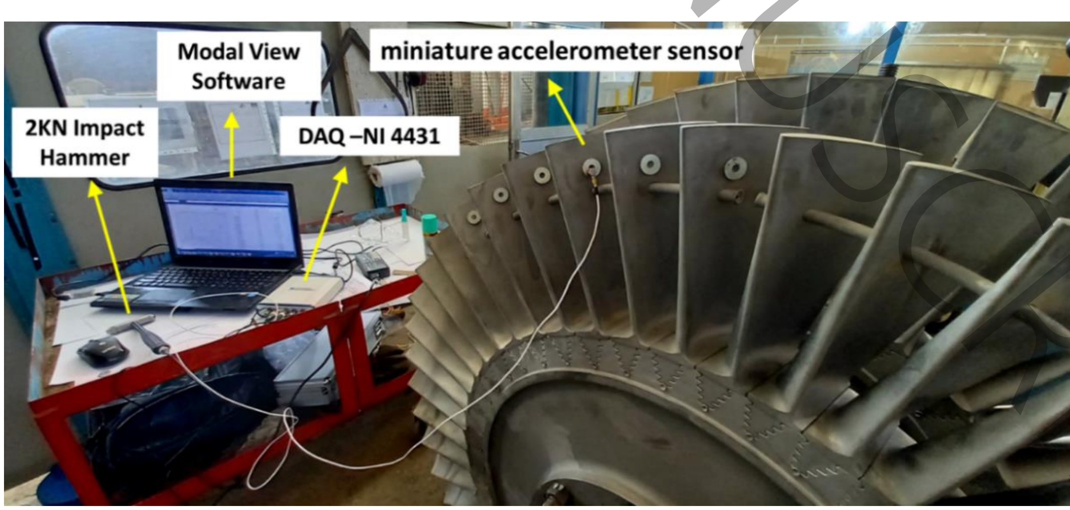

Fig. 13. Group blades assembly Vibration test setup

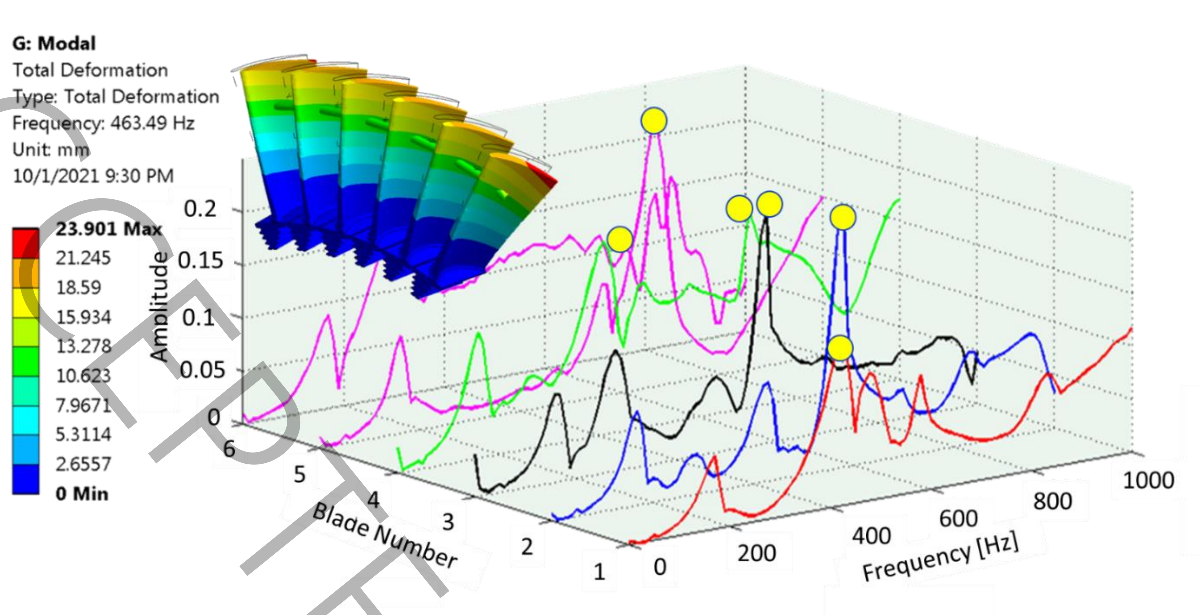

Fig. 14. Comparison FEM and test results of 1st natural frequency for group blades

The measured value of the frequency for the group-blades is about 430 Hz (yellow points) and the value calculated by the FEM is 463.5 Hz. The 6% difference between the modal test and the FEM shows the high accuracy of modeling, definition of boundary conditions and material.

As stated in Section 3.1, the modal parameters of a single blade are different from a group-blades coupled by a damping tube. Therefore, the modal characteristics of a group-blades system cannot be accurately predicted by a single blade analysis. In this section, the comparison of the results of the natural frequencies and the mode shapes of the single blade and the group-blades coupled by the damping tube has been investigated. The purpose of comparing the modal characteristics of a single blade and group-blades is to find the answer to the following two questions:

1- What are the differences between the dynamic behavior of a single blade and group-blades, and what was the philosophy of the designer?
2- Considering that the damping tube is damaged under the effect of erosion, what changes will there be in the critical modes and the dynamic stresses?

In order to investigate the causes of the failure of the group-blades, it is first necessary to evaluate the design philosophy of the manufacturer. In the first step, the stress and vibrations of a single blade are analyzed. Single-blade boundary conditions are the condition of a fixed support in fir tree root and the loading conditions are the effects of temperature, pressure and centrifugal force distribution due to rotational speed.

The Campbell diagram for a single blade and a group of blades coupled to each other with a damping tube is shown in Fig. 15 and Fig. 16. The Campbell diagram is an overall view of regional vibration excitation that can occur on an operating system. The Campbell diagram can be generated from machine design criteria or from machine operating data.

For the single-blade mode, which somehow indicates the lack of connection of the damping tube between the blades, two natural frequency interferences with excitation harmonics have occurred in the working speed range. One of the interfere is related to the first bending mode with the fifth harmonic and the other interference is related to the sixth mode with the 42nd harmonic, which is related to the passing frequency of the second stage stator blades.

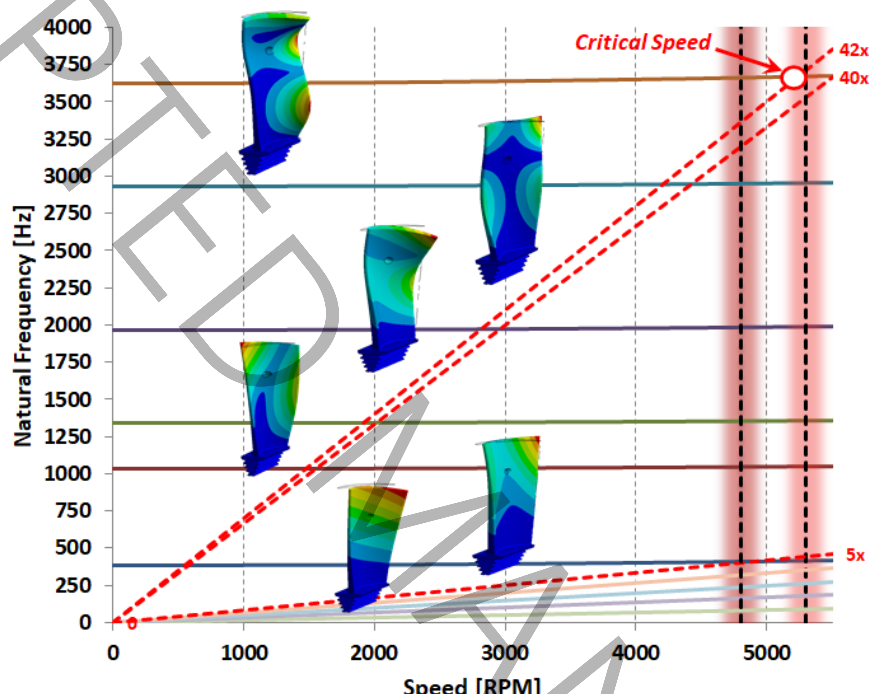

Fig. 15. Campbell diagram of single blade

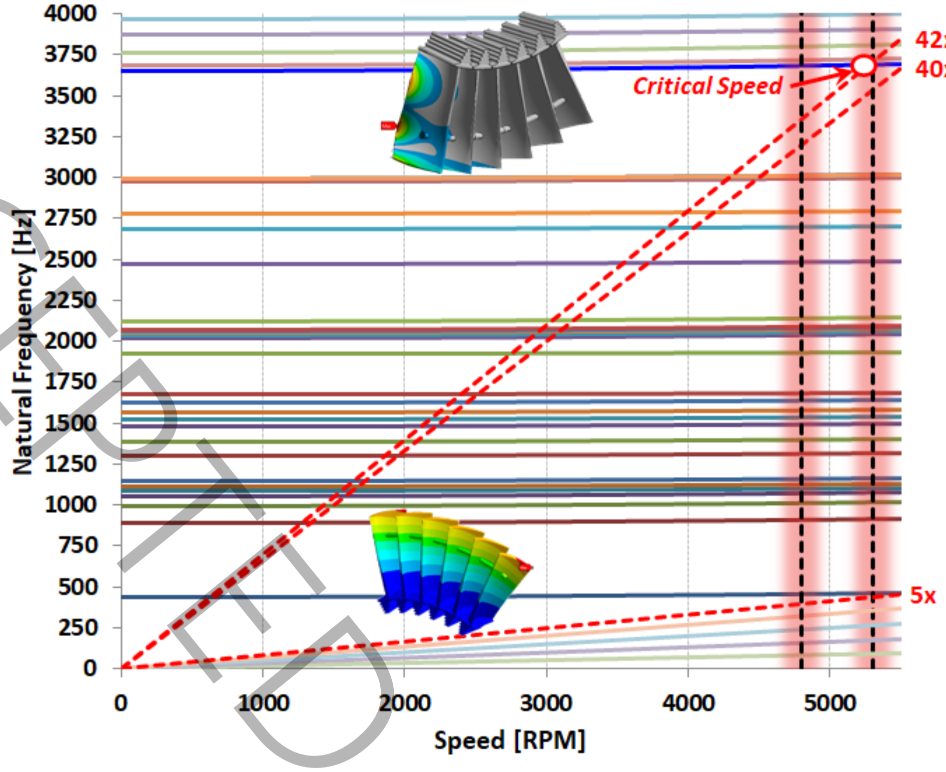

Fig. 16. Campbell diagram of group blades

Since the turbine blades vibrate due to the low harmonics of the engine, these excitations should also be considered in the Campbell diagram. Campbell's diagram for the first mode is shown in Fig. 17. Damage in the damping tube causes the connection between the vanes to break. So that a single blade will have a lower frequency than the group of blades. The results of the diagram show that the cutting of the tube causes a 30% decrease in the frequency of the first bending mode of the blade in such a way that it will interfere with the fifth harmonic of the excitation in the range of 4900 rpm.

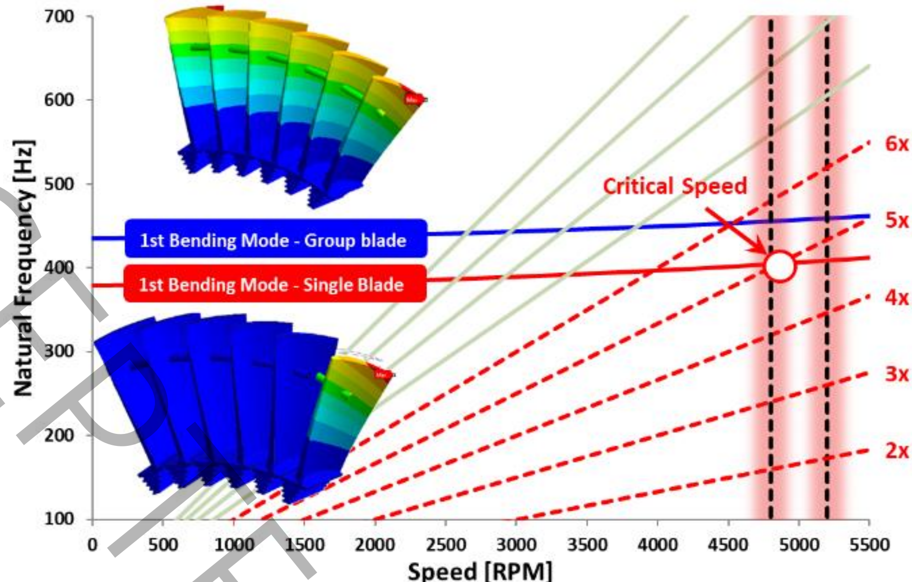

Fig. 17. Comparison of Campbell's diagram of the first mode of the blade group in the state of healthy and damaged tube

Comparing the Campbell diagram of the group of blades in healthy and damaged conditions, it is clear that the cut of the connection between the blades will lead to the resonance of the blade in the first mode with the fifth harmonic.

In the following, checking whether the initial design has been done correctly by the turbine manufacturing company (OEM), the sensitivity of the existing system is evaluated. The position of the tube installation and the internal diameter of the tube on the vibrations of the first bending mode of the blade are shown in the

Fig. 18 and Fig. 19.

Based on Fig. 18 the distance of the tube to about 70% of the hub blade increases the frequency of the first bending mode and for a distance of more than 70% reduces the natural frequency. The main reason for the decrease in blade natural frequency at the tube position in the last 30% is the effect of the mass of the tube at the blade tip or the predominant effect of the concentrated mass relative to the stiffness effect.

Fig. 19 shows increasing the diameter ratio reduces the stiffness of the connection between the blades. Reducing the difference between the inner and outer diameters reduces the natural frequencies of the blades. The best design is the diameter ratio of 0.6, which is in the safe range of resonance for the first mode.

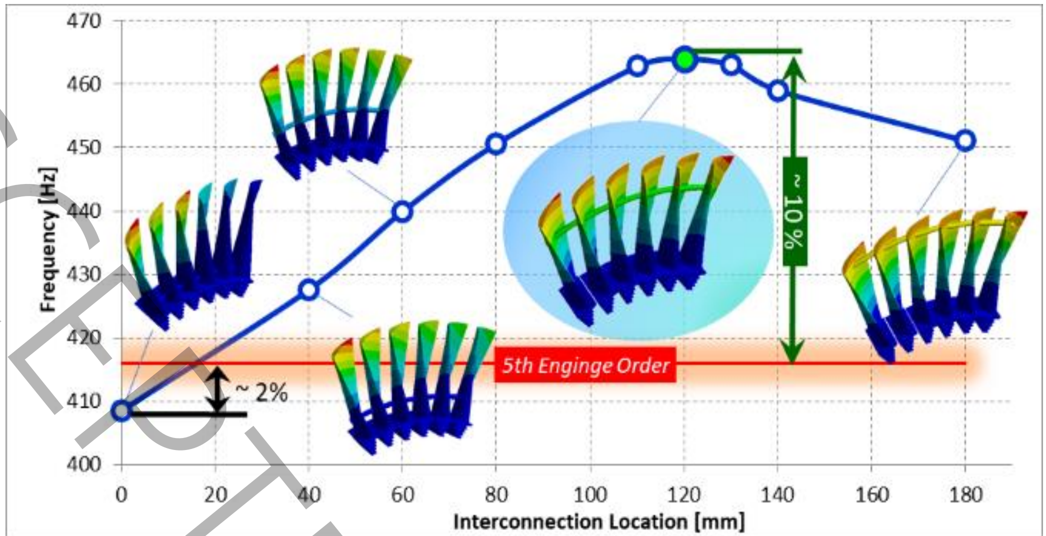

Fig. 18. The 1st bending natural frequency versus interconnection location

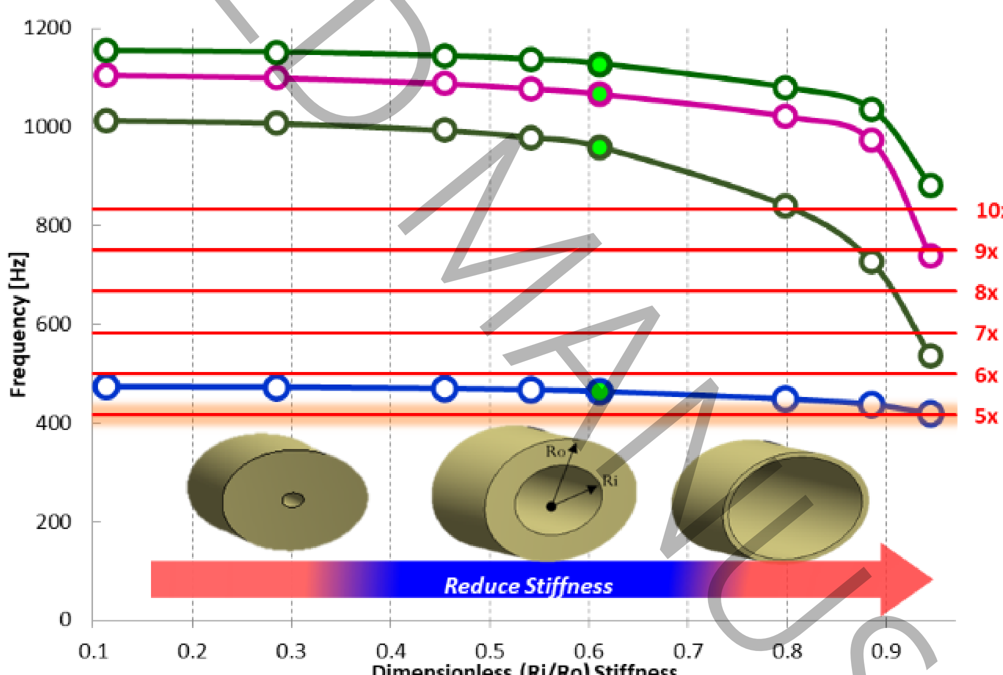

Fig. 19. The 1st four natural frequencies versus interconnection stiffness

Based on the analyzed sensitivity, it can be concluded that the main design criterion of the OEM is to avoid the resonance of the first bending mode of the blade with low order harmonics of the turbine. For this purpose, harmonic analysis is conducted in order to obtain the dynamic stresses of critical modes. The first step to obtain dynamic stresses is the accurate calculation of damping coefficients.

In the following, the dynamic stresses of the blade are calculated in the state of healthy and damaged damping tubes. Dynamic stress is performed in the harmonic response environment of ANSYS software. In order to accurately analyze the stress and amplitude of vibrations, it is necessary to extract the damping ratios of the system using the modal test.

The damping ratios of blades from impact test measurement for single blade and groups blades are both calculated by the half power bandwidth method. The principle of the half-power bandwidth method is that we need to obtain the resonance peak of frequency response of the system, and then damping parameters of the system are obtained by using the resonance frequency and two frequency points with 1/√2 times decrease in the resonance amplitude near resonance peak, so the solution formula can be expressed as following[35].

$$2\xi = \frac{f_2 - f_1}{f_0}$$

The ξ is the damping ratio, f_0 resonance frequency, f_1 and f_2 the two frequency values which are 1/√2 times resonance peak at the two sides of resonance frequency as shown in Fig. 20 for single blade and Fig. 21 for group-blades. The damping ratio calculated by the half-power method for a single blade is 0.006 and 0.058 for a group of blades. The damping tube has increased the damping ratio by ten times. Analyzing the effects of the damping ratio on dynamic stresses and vibration amplitude is presented in the harmonic analysis section.

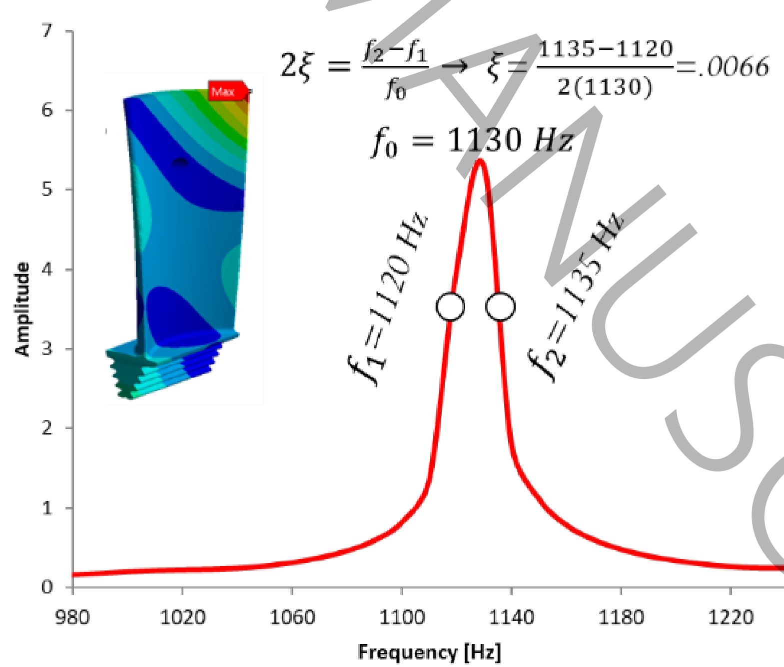

Fig. 20. Damping Ratio of single blade

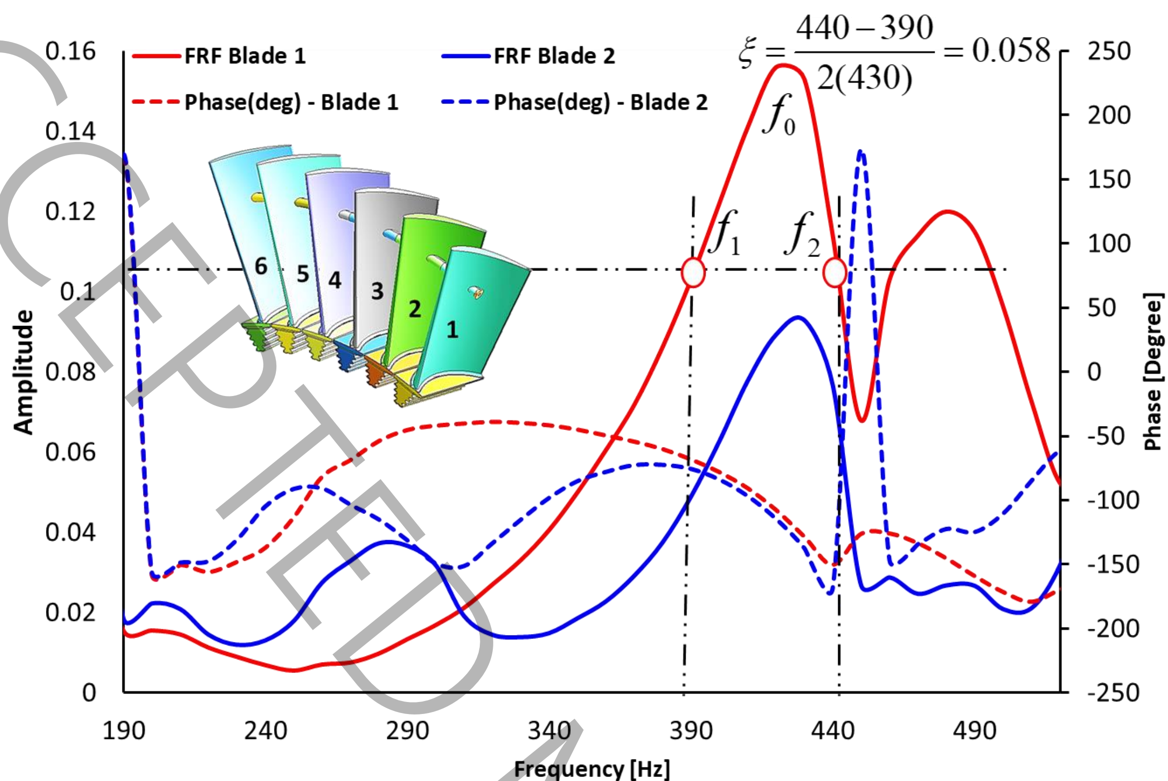

Fig. 21. Damping ration of group blades

When the damping tube is failed, one blade will lose its connection with other blades. This leads to a decrease in the natural frequency of a single blade compared to connected blades. According to Fig. 22, the single blade will vibrate with a frequency of 400 Hz with an amplitude of 1.1 mm and a dynamic stress of ± 38 MPa. If the group of blades connected to each other have a frequency of about 450 Hz and the range of vibrations is about 0.6 mm and the dynamic stress is ± 5 MPa. As a result, the presence of the damping tube between the blades will help to control the range of vibrations and reduce dynamic stresses in addition to increasing the natural frequency and staying away from the critical area. Reducing the dynamic stress will increase the fatigue life of the blades.

Also, the dynamic stresses in the first mode for a single blade and a group-blades are shown with red and blue dashed lines in the Fig. 22, respectively. The place of stress concentration in the blade fir tree root will be prone to high cycle fatigue cracks in the first mode in case of resonance.

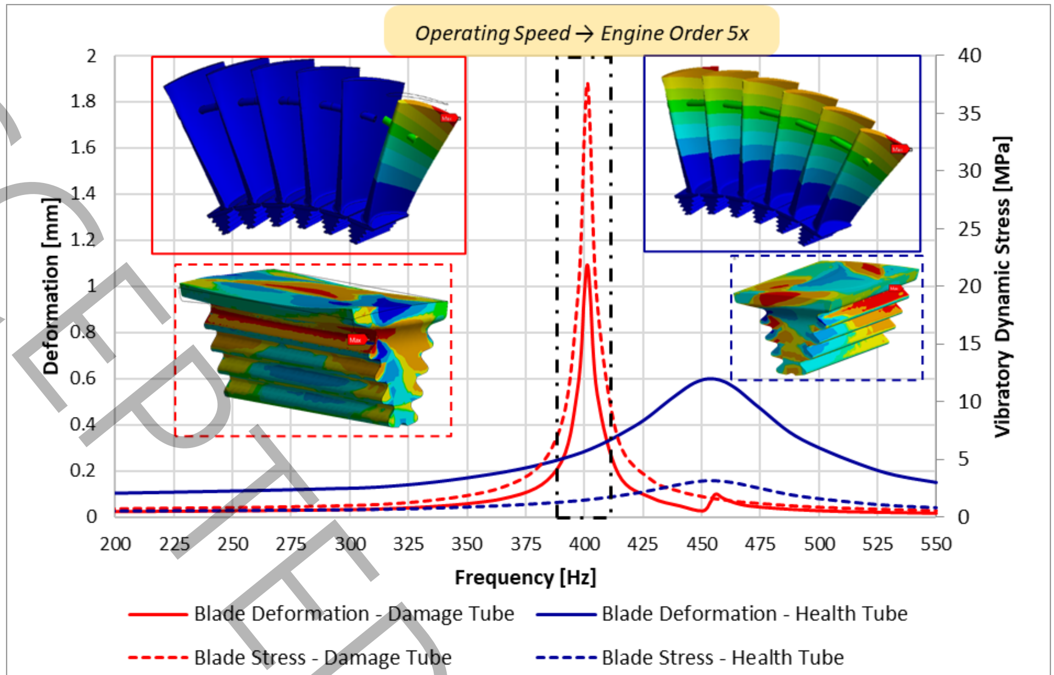

Fig. 22. The graph of frequency response for 1st bending mode with the excitation fifth harmonic per-revolution of rotor.

The frequency response diagram for the frequency of the 6th mode, which resonates under the effect of the 42nd harmonic, is shown in the Fig. 23. A single blade is shown with a red line and a group blade is shown in blue line. In general, the amplitude in this mode is very low, and it seems that it is not a critical mode from the point of view of the OEM.

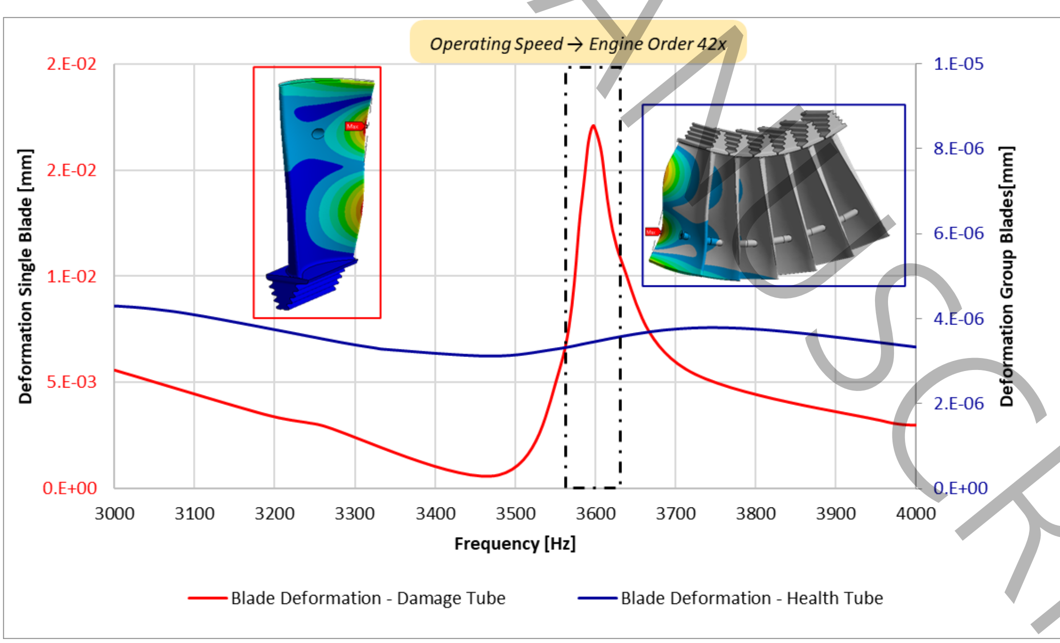

Fig. 23. The graph of frequency response for 6th bending mode with the excitation fifth harmonic per-revolution of rotor.

Conclusion

Failure analysis of a gas turbine group blade that connected with damping tube under operation about 40 thousand hours was performed. The blades and damping tube were made by Waspaloy and IN718. The following results were obtained from this investigation.

1. According to visual inspection, wear effects were observed on the surface of the damping tube.
2. Microscopic images of the fracture section of the damping tube with 60x magnification show the effect of the pit on the outer surface of the damping tube. Around the pit, microcracks and the beach mark area, which is caused by fatigue crack growth, were observed. The presence of pits on the surface of the tube can create points with high dynamic stress concentration, which in combination with the mean stress under static loads will lead to fatigue crack growth and tube failure. As a result, the failure factor of the damping tube is a combined phenomenon called Erosion-Fatigue.
3. The static stress analysis shows that in the case where the damping tube is healthy, the Von Mises equivalent stress is 561 MPa. In the event of erosion on the surface of the damping tube, the Von Mises stress increases up to 963 MPa. Considering that the yield stress of the IN718 alloy is 1027 MPa, it can be concluded that the pitting caused by erosion is the main cause of the failure of the damping tube.
4. In the following, checking whether the initial design has been done correctly by the turbine manufacturing company (OEM), the sensitivity of the existing system is evaluated. The position of the tube installation and the internal diameter of the tube on the vibrations shows that the designer has considered the best choice to avoid resonance of the first bending mode.
5. Cutting the damping tube causes a change in the boundary conditions of the blade and, as a result, a change in the natural frequencies, which has caused a 30% decrease in the frequency of the first bending mode of the blade. Reducing the desired frequency leads to the occurrence of blade resonance under the effect of the fifth harmonic excitation force.
6. The dynamic stress calculations show that the first bending mode is the most critical mode. If the damping tube is damaged, the natural frequency of the first bending mode of the blade will decrease by 30% and will be exposed to resonance, which will result in a 2 times higher amplitude and 9 times higher dynamic stress compared to a healthy damping tube.

Acknowledgments: The authors wish to express their gratitude to Mr. Askari and Mr. Behravan from NISOC, and also to Dr. Darvish for their support.

References

[1] M. Schubert, J. Tusche, Development of a Robust LP Blade Family for Variable Speed Applications, in: Turbo Expo: Power for Land, Sea, and Air, American Society of Mechanical Engineers, 2018, pp. V008T029A030.

[2] J. Szwedowicz, T. Secall-Wimmel, P. Dünck-Kerst, Damping performance of axial turbine stages with loosely assembled friction bolts: The nonlinear dynamic assessment, Journal of Engineering for Gas Turbines and Power, 130(3) (2008).

[3] R. Drozdowski, L. Völker, M. Häfele, D. Vogt, Experimental and numerical investigation of the nonlinear vibrational behavior of steam turbine last stage blades with friction bolt damping elements, in: Turbo Expo: Power for Land, Sea, and Air, American Society of Mechanical Engineers, 2015, pp. V008T026A007.

[4] J. Voldřich, J. Lazar, P. Polach, Nonlinear Vibration Analysis of Steam Turbine Rotating Wheel Equipped with the LSB48 Blades, in: The 14th IFToMM World Congress, 2015, pp. 48-55.

[5] E. Ferhatoglu, S. Zucca, D. Botto, J. Auciello, L. Arcangeli, Nonlinear Vibration Analysis of Turbine Bladed Disks With Mid-Span Dampers, in: Turbo Expo: Power for Land, Sea, and Air, American Society of Mechanical Engineers, 2020, pp. V011T030A020.

[6] E. Petrov, D. Ewins, Effects of damping and varying contact area at blade-disk joints in forced response analysis of bladed disk assemblies, (2006).

[7] D. Charleux, C. Gibert, F. Thouverez, J. Dupeux, Numerical and experimental study of friction damping blade attachments of rotating bladed disks, International Journal of Rotating Machinery, 2006 (2006).

[8] E. Petrov, D. Ewins, Analytical formulation of friction interface elements for analysis of nonlinear multi-harmonic vibrations of bladed disks, J. Turbomach., 125(2) (2003) 364-371.

[9] C. Siewert, L. Panning, J. Wallaschek, C. Richter, Multiharmonic forced response analysis of a turbine blading coupled by nonlinear contact forces, Journal of Engineering for Gas Turbines and Power, 132(8) (2010).

[10] D. Laxalde, F. Thouverez, J.-P. Lombard, Forced response analysis of integrally bladed disks with friction ring dampers, Journal of Vibration and Acoustics, 132(1) (2010).

[11] S. Zucca, C.M. Firrone, M. Facchini, A method for the design of ring dampers for gears in aeronautical applications, (2012).

[12] K.Y. Sanliturk, D.J. Ewins, A. Stanbridge, Underplatform dampers for turbine blades: theoretical modeling, analysis, and comparison with experimental data, J. Eng. Gas Turbines Power, 123(4) (2001) 919-929.

[13] E. Cigeroglu, N. An, C.-H. Menq, Forced response prediction of constrained and unconstrained structures coupled through frictional contacts, Journal of Engineering for Gas Turbines and Power, 131(2) (2009).

[14] S. Zucca, C.M. Firrone, M. Gola, Modeling underplatform dampers for turbine blades: a refined approach in the frequency domain, Journal of Vibration and Control, 19(7) (2013) 1087-1102.

[15] M. Klein, G. Simpson, The development of innovative methods for erosion testing a Russian coating on GE T64 gas turbine engine compressor blades, in: Turbo Expo: Power for Land, Sea, and Air, 2004, pp. 347-352.

[16] A. Kochetov, D. Korzh, A. Egorov, Lowering the erosion wear of long blades of the last stages by optimum hanging, Power Technology and Engineering, 41(1) (2007) 41-46.

[17] G.I. Ilieva, Erosion failure mechanisms in turbine stage with twisted rotor blade, Engineering Failure Analysis, 70 (2016) 90-104.

[18] M.S. Khan, C. Sasikumar, A water droplet erosion-induced fatigue crack propagation and failure in X20Cr13 martensitic stainless-steel turbines working at low pressure, Engineering Failure Analysis, 139 (2022).

[19] Y. Mollapour, E. Poursaeidi, O. Pedram, Study of pitting corrosion under actual operating conditions of a first stage compressor blade, Engineering Failure Analysis, 131 (2022).

[20] P. Yang, W. Yue, J. Li, G. Bin, C. Li, Review of damage mechanism and protection of aero-engine blades based on impact properties, Engineering Failure Analysis, 140 (2022) 106570.

[21] D. Qi, P. Petrie-Repar, T. Gezork, T. Sun, Establishment of an open 3D steam turbine flutter test case, in: 12 th European Conference on Turbomachinery Fluid dynamics & Thermodynamics, EUROPEAN TURBOMACHINERY SOCIETY, 2017.

[22] C. Fuhrer, D.M. Vogt, On the impact of simulation approaches on the predicted aerodynamic damping of a low pressure steam turbine rotor, in: Turbo Expo: Power for Land, Sea, and Air, American Society of Mechanical Engineers, 2017, pp. V008T029A007.

[23] R. Rzadkowski, V. Gnesin, L. Kolodyazhnaya, Aeroelasticity analysis of unsteady rotor blade forces and displacements in LP last stage steam turbine with various pressure distributions the stage exit, Journal of Vibration Engineering & Technologies, 6 (2018) 333-337.

[24] P. Wang, X. Wu, X. He, Vibration-Theoretic Approach to Vulnerability Analysis of Nonlinear Vehicle Platoons, IEEE Transactions on Intelligent Transportation Systems, (2023).

[25] D. Graciano, J. Rodríguez, G. Urquiza, M. Tecpoyotl-Torres, Damage evaluation and life assessment of steam turbine blades, Theoretical and Applied Fracture Mechanics, 124 (2023) 103782.

[26] Y. Wang, D. Luo, Y. Huang, Y. Peng, X. Wang, X. Yan, H. Xiao, Z. Huang, Nonlinear dynamic model of a turbine blade considering vibration and crack coupling, Journal of Theoretical and Applied Mechanics, 62(1) (2024) 75-88.

[27] P. Hu, Q. Meng, W. Fan, W. Gu, J. Wan, Q. Li, Vibration characteristics and life prediction of last stage blade in steam turbine Based on wet steam model, Engineering Failure Analysis, (2024) 108127.

[28] X. Kan, B. Zhao, A New Method of Reducing the Vibration Localization of the Multi-packet Whole Mistuned Bladed Disk System Using Bladed Packets, International Journal of Aeronautical and Space Sciences, 22(4) (2021) 857-865.

[29] A. Chatterjee, Lumped parameter modelling of turbine blade packets for analysis of modal characteristics and identification of damage induced mistuning, Applied Mathematical Modelling, 40(3) (2016) 2119-2133.
[30] A. Chatterjee, M.S. Kotambkar, Modal characteristics of turbine blade packets under lacing wire damage induced mistuning, Journal of Sound and Vibration, 343 (2015) 49-70.
[31] G. Ma, M. Xu, L. Dong, Z. Zhang, Multi-point suspension design and stability analysis of a scaled hoop truss antenna structure, International Journal of Structural Stability and Dynamics, 21(06) (2021) 2150077.
[32] X. Kan, K. Wang, B. Zhao, Dynamic characteristics of vibration localization of mistuned bladed disk due to shroud and blade damages, Journal of Low Frequency Noise, Vibration and Active Control, 43(1) (2024) 405-417.
[33] Chapter 10 - Advances and Innovations in LNG Industry, in: S. Mokhatab, J.Y. Mak, J.V. Valappil, D.A. Wood (Eds.) Handbook of Liquefied Natural Gas, Gulf Professional Publishing, Boston, 2014, pp. 437-464.
[34] B. Salehnasab, E. Poursaeidi, S.A. Mortazavi, G.H. Farokhian, Hot corrosion failure in the first stage nozzle of a gas turbine engine, Engineering Failure Analysis, 60 (2016) 316-325.
[35] J. Zhang, P. Shan, K. Cheng, D. Ye, Comparison of Blade Tip Timing With Strain Gauge Data for Evaluation of Dynamic Characterization of Last Stage Blade With Interlocked Shroud for Steam Turbine, in: ASME Turbo Expo 2018: Turbomachinery Technical Conference and Exposition, 2018.